

Data Based Cylinder Pressure Modeling for Direct-injection Diesel Engines

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ABSTRACT

In this article a new zero-dimensional model is presented for simulating the cylinder pressure in direct injection diesel engines. The model enables the representation of current combustion processes considering multiple injections, high exhaust gas recirculation rates, and turbocharging. In these methods solely cycle-resolved, scalar input variables from the electronic control unit in combination with empirical parameters are required for modeling. The latter are adapted automatically to different engines or modified applications using measured cylinder pressure traces. The verification based on measurements within the entire operating range from engines of different size and type proves the universal applicability and high accuracy of the proposed method.

INTRODUCTION

The cylinder pressure trace provides a superior feedback signal for the combustion process. It allows for accurate calculation of corresponding quality criteria and the design of closed loop control strategies [1, 2]. Since the development of such functions depends on a multitude of time- and cost-intensive test bench measurements, the application of combustion models is of great interest within the software development for the electronic diesel control (EDC).

In this context especially zero-dimensional models come into consideration due to their simple formulation and short calculation time. They can be used to support the function development in offline-simulation, whereas the

potential application areas range from the test of function prototypes to controller design, robustness analysis and software-in-the-loop tests of EDC code. Thereby the verification and validation of the developed concepts becomes possible in early stages of development.

Since the use of pressure sensors in mass-production vehicles is still affected by challenges such as durability and costs, the importance of simulation based approaches has also increased for online applications in the EDC unit. To address these issues, zero-dimensional pressure models offer the potential of a real-time implementation in terms of a virtual sensor for the engine management. The possible applications range from the estimation of combustion parameters to engine diagnostics functions and optimization of the injection process.

In the following a short review of existing zero-dimensional approaches for combustion modeling is given. The group of empirical models applies a parameterized heat release rate, also known as the equivalent heat release, which is generally based on the well-known Vibe function [3, 4]. In a second step, the model parameters are described in dependency of engine control variables by means of comparatively simple polynomial and power functions. For this purpose, series of measurements with the variation of single parameters based on a reference point, also known as the "one change at a time"-method, are utilized. Known representatives of this class are the models of Woschni and Anisits [5], Oberg, Schreiner and Barba [6].

Besides also phenomenological approaches are used for heat release modeling. These models perform a simplified description of basic physical and chemical processes like spray dispersion, evaporation, fuel-mixture generation and reaction kinetics based on the injection rate and further cycle-resolved process variables [6]. With regard to the intended applications in software development and engine control only the comparatively simple models with a global description of the dominant rate-determining processes come into question. Since no quasi-dimensional division in different zones is performed, local conditions cannot be dissolved.

One example is the model from Constien [4, 7] which is based on the assumption, that the fuel combustion is basically governed by the ignition delay and the fuel evaporation. The model from Chmela et al [8] utilizes the concept of mixing controlled combustion. It states the essential assumptions that the burn rate is proportional to the currently available fuel mass and the conversion speed depends on the local density of the turbulent kinetic energy. The approach was advanced by Pirker et al [9]. Barba proposed a more detailed phenomenological model [10], which accounts also for special features of current combustion processes like pilot injections and high exhaust gas recirculation. It comprises submodels for premixed and diffusion combustion supplemented by models for fuel evaporation and ignition delay.

Another established approach in the field of engine modeling is the application of advanced data based modeling techniques, which has also been addressed repeatedly in literature. He and Rutland use multilayer-perceptron (MLP) networks to represent the cylinder pressure trace as a function of injection mass and engine load, whereas data from CFD-simulation was used for training [11]. Wang et al follow the approach to build separate models for different cylinder pressure features like maximum pressure and maximum rising rate position on the basis of MLP networks [12]. Eventually Winsel et al applied time-delay neural networks for modeling the cylinder pressure of a SI-engine [13]. The activation functions of the perceptrons were used for the functional representation of the pressure trace.

With regard to the aspired applications the essential prerequisite is a model which is not only able to reproduce qualitative trends but also calculates correct absolute values within the entire operating range accompanied by low resource demands and fast adaption to different engines. On the other hand, a forecast and optimization of the thermodynamic combustion process without prior parameter adaption is not claimed. In this context the benefit of the available zero-dimensional combustion models is limited. Whereas the empirical approaches offer the possibility to examine the influences of single parameters, the accuracy drops with increasing distance from the reference point. Furthermore the model adaption is

comparatively involved since extensive "one-change-at-a-time" measurements are necessary.

The advantage of phenomenological models lies in the physical background, which allows extrapolation to a certain degree. While building up on the injection rate, they are particularly suitable for the execution of parameter studies and combustion process development. However, both the computation time for model evaluation as well as the effort for parameter identification is increased since the calculation is carried out by solving differential equations. The identification of a joint parameter set for the whole operation range along with the achievement of sufficient crank-angle resolved accuracies is difficult according to experience. In order to meet the requirements of actual combustion processes like post injections, extensions to the individual model structures are necessary [14]. Furthermore it is noted, that these approaches focus on the representation of the (gross) heat release rate, which has to be converted to the pressure curve by performing a heat release analysis. This involves an increased computational load compared to an explicit pressure model, which complicates online-applications. In addition further uncertainty factors such as the utilized heat transfer and leakage models are introduced.

In order to comply with the stated requirements the development of a new model becomes necessary. The chosen approach comprises a coupling of data based modeling techniques with a parametric cylinder pressure representation. The basic cornerstones can be summarized as follows:

1. Choice of an empirical model structure using only cycle-resolved input data without incorporation of the injection rate
2. Direct representation of the pressure curve by using parametric relationships to allow an algebraic calculation without the need of a heat release analysis
3. Development of robustness and timely methods for automated identification of model parameters (e.g. ignition delay) from measured pressure traces
4. Use of data based modeling techniques to enable fast parameter identification from standard engine measurements along with high modeling accuracies
5. Continuous validation and verification of all algorithms and approaches with measurement data from the entire operating range of different engines

The paper is organized as follows: After a short explanation regarding the procedures of data collection and pre-processing the model structure along with the submodels for compression pressure, ignition delay and combustion pressure are introduced. At the same time the respective methods for parameter identification are explained. On this basis the scheme for coupling the submodels to the overall cylinder pressure model is developed. The applied data based modeling technique as well as the algorithm for selecting the input parameters are illustrated next.

model validation an overview of the analyzed engines and datasets is given in the following. Finally the results are presented and discussed in the concluding section.

DATA COLLECTION AND PREPROCESSING

The cylinder pressure traces were measured with water-cooled piezoelectric test bench sensors [15] and pre-processed according to the following steps:

1. Offset compensation of every pressure cycle by referencing the pressure values in the range of "intake closes" to the intake manifold pressure
2. Averaging of pressure traces of all cycles
3. Low pass filtering to prevent flow pipe waves
4. Averaging of averaged pressure traces of all valid cylinders: Since the pressure model cannot account for cylinder-wise differences, it is trained on an averaged cylinder pressure, which gives the best representation of the combustion characteristic. Cylinders with signal distortions or identified bias of the pressure sensor are not considered for averaging.
5. Filtering of operating points with unstable combustion: For this purpose the compression pressure for every operating point is estimated and subtracted from the pressure traces of all cycles. The filtering criterion is formed from the variation of the maximum difference pressure.

The calculations are carried out for all integral crank angles of the high pressure phase within the range of closed valves. For transient measurements all cycles are analyzed individually and according to this, steps 2, 4 and 5 are skipped. Before applying data based modeling techniques all data is normalized to zero mean and a variance of unity to obtain better numerical conditioning.

MODEL STRUCTURE

The applied approach comprises a separation of the measured cylinder pressure trace in a compression and combustion part and subsequent modeling with parametric models. In these methods, the contributions of pilot, main and post injections are considered separately, using appropriate shape functions according to the determined starts of combustion. In this context, a multitude of operating points have been analyzed to develop a universally applicable model representation for typical compression and combustion pressure traces of current combustion processes. In summary, the developed cylinder pressure model consists of three coupled submodels for compression pressure, combustion pressure and ignition delay. The different pressure components are herein referred to as stated in Table 1.

COMPRESSION PRESSURE – The compression pressure equals the cylinder pressure if no combustion takes place. Due to heat loss and leakage effects, it differs significantly from the ideal polytropic curve. As starting point, pressure curves in motored condition were analyzed. However, the mentioned loss effects are only

Table 1: Overview pressure components

p_{mot}	compression pressure in motored condition (no injection, no combustion)
p_{comp}	compression pressure (not including a combustion event, but not necessarily in motored condition)
$p_{\text{comp,ext}}$	extrapolated compression pressure
p_{cmb}	combustion pressure, $p_{\text{cmb}} = (p_{\text{Pil}}) + p_{\text{MI}} + (p_{\text{Pol}})$
p_{meas}	measured total pressure, $p_{\text{meas}} = p_{\text{comp}} + p_{\text{cmb}}$

partially observable in this operating mode, since engine parameters like boost pressure and intake manifold temperature can only be varied in a limited way. For identification of the relevant input variables for modeling the compression, further measurements in fired condition have been conducted, whereas the injection was shut off in one cylinder. Besides the intake manifold pressure p_{IM} with an almost proportional influence on the maximum pressure p_{max} , a distinct dependency from the engine speed n exists due to the time-dependency of the loss effects ($d\varphi \sim n \cdot dt$). Furthermore the condition of the cylinder charging affects the compression, which can be captured approximatively by p_{IM} and the intake manifold temperature T_{IM} . The latter is closely linked to the exhaust recirculation rate r_{EGR} , which is defined as the ratio between the recirculated gas mass m_{EGR} to the total in-cylinder gas mass m_{gas} . Figure 1 illustrates the dependencies.

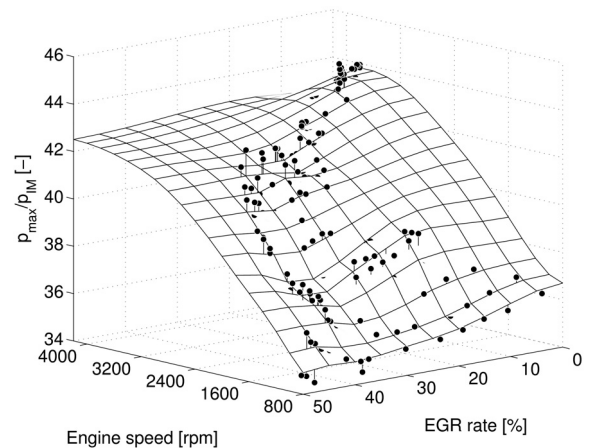


Figure 1: Dependencies of the normalized maximum compression pressure

After analyzing the most important influencing factors the task consists in the formulation of a suitable model representation for the asymmetric progression of the compression pressure trace. For this purpose, an approach based on a nonlinear polynomial model $\hat{p}_{\text{mot}} = \hat{\theta}Z^T$ is used. The columns of the regressor matrix $Z \in \mathbb{R}^{N \times n}$ represent the n regressors, which comprise combinations of the $m+1$ input variables $\{u_1, \dots, u_m, 1\}$ up to a determined order. Here N denotes the number of considered crank angle values. In this connection u_1 is chosen as a crank angle dependent basis function and $\{u_2, \dots, u_m\}$ are further

cycle resolved engine parameters such as crankshaft speed or intake manifold temperature. Additionally, shiftings of the basis function u_1 up to the integer crank angle value $d_u \in \mathbb{N}$ are considered in $\mathbf{Z}$. According to this the modeled value for motored pressure at crank angle φ_0 depends on:

$$\hat{p}_{\text{mot}}(\varphi_0) = f(u_1(\varphi_0), \dots, u_1(\varphi_0 - d_u), \dots, u_2(\varphi_0), \dots, u_m(\varphi_0)) \quad (1)$$

Suitable shape functions u_1 are for example the following approximations of the compression pressure:

- adiabatic model ($\kappa = \text{const} = 1.4$)

$$p_{\text{ad}}(\varphi) = p_{\text{IM}} \left(V(\varphi_{\text{IC}}) / V(\varphi) \right)^{\kappa} \quad (2)$$

where: $V(\varphi)$: volume function of engine

φ_{IC} : crank angle, where intake valve closes

- polytropic model [3]

$$dp_{\text{poly}} = -\frac{1}{V} \kappa(p_{\text{poly}}, T_{\text{poly}}, \lambda) p_{\text{poly}} dV \quad (3)$$

The tables for caloric data $\kappa = f(p_{\text{poly}}, T_{\text{poly}}, \lambda)$ are taken from DeJaegher [3].

- Single-zone thermodynamic model [3]

$$dp_{\text{th}} = -\frac{1}{V} (\kappa(p_{\text{th}}, T_{\text{th}}, \lambda) p_{\text{th}} dV + (\kappa - 1) dQ_{\text{ht}}) \quad (4)$$

For determining the heat transfer dQ_{ht} to the chamber walls, the empirical correlation from Hohenberg for convective heat transfer is used [16].

Since the number of potential regressors increases rapidly with higher orders and input dimensions, the application of suitable methods for reducing the model complexity is necessary. For this purpose a heuristic wrapper-algorithm with stepwise structure selection is used, which involves repeated model evaluations with different regressor sets (Figure 2, [17]).

At first, an initial model with a single regressor is identified, which minimizes the RMSE-criterion. In a subsequent selection step, the next regressor is searched, which leads to a maximum RMSE-reduction in combination with the already selected regressor set. Thereupon every selected regressor is tentatively deleted in a pruning step. Through this, it is checked if a combination of regressors exists, which allows for a lower RMSE while having the same number of selected inputs. The process is terminated if the relative minimization of the RMSE is below a given threshold. In each step the parameters $\hat{\theta}$ are calculated with the least-squares method (Eq. 5).

$$\hat{\theta} = (\mathbf{Z}^T \mathbf{Z})^{-1} \mathbf{Z}^T \mathbf{p}_{\text{comp}} \quad (5)$$

Motored condition – The model construction is carried out in two steps. At first, the measured pressure traces in motored condition are modeled in the described manner in dependency of a basis function, the engine speed and the intake manifold pressure.

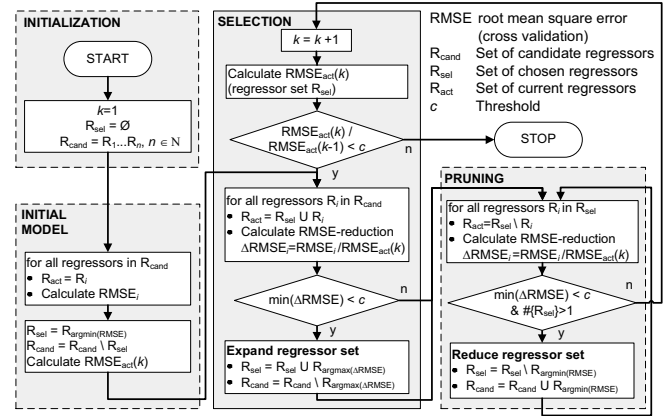


Figure 2: Stepwise wrapper algorithm

Figure 3 compares the absolute errors of different modeling approaches for 14 motored pressure traces. In this example the normalized adiabatic shape function $p_{\text{ad}^*} = p_{\text{ad}} / p_{\text{IM}}$ is used for the polynomial model since it allows for an algebraic calculation. Whilst the modeling error of the conventional approaches is unacceptable, the proposed method allows an excellent adaption. Only by shifting the basis function, the representation of the asymmetric compression curve with a symmetric shape function like the adiabatic curve becomes possible. Eq. 6 shows exemplarily the determined correlation for the model shown in Figure 3.

$$p_{\text{mot}}(\varphi) = p_{\text{IM}} \hat{\theta}_{\text{mot}} \mathbf{Z}_{\text{mot}}^T \quad (6)$$

$$\mathbf{Z}_{\text{mot}} = \begin{bmatrix} p & pn & pn^2 & p^2 & pp_{-1} & p_{-1} \end{bmatrix}$$

where: $p = p_{\text{ad}^*}(\varphi)$, $p_{-1} = p_{\text{ad}^*}(\varphi - 1)$

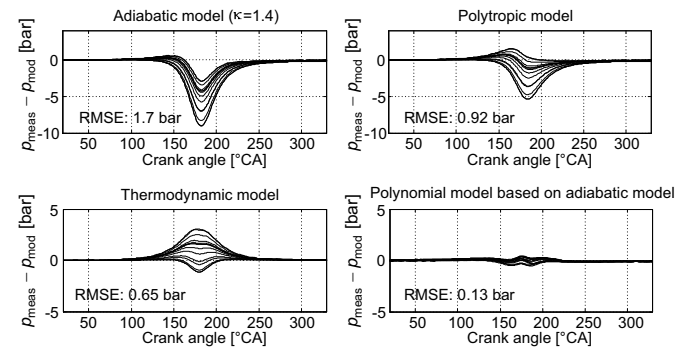


Figure 3: Modeling of motored pressure traces with different approaches (Engine PC 1a, cf. Tables 3 and 5)

Fired condition – Subsequently the compression pressure is extrapolated for every operating point in fired condition by adjusting the modeled motored pressure through scaling and shifting in the crank angle area up to the first start of combustion $\varphi_{\Delta p}$ (Figure 4). The parameters are again determined by least-squares (Eq. 7, 8).

$$\hat{\theta}_{\text{ext}} = (\mathbf{Z}^T \mathbf{Z})^{-1} \mathbf{Z}^T \mathbf{p}_{\text{meas}}; \mathbf{Z} = [\mathbf{p}_{\text{mot}} \mathbf{1}] \forall \varphi \in [\varphi_{\text{IC}} \varphi_{\Delta p}] \quad (7)$$

$$\mathbf{p}_{\text{comp,ext}} = \hat{\theta}_{\text{ext,1}} \mathbf{p}_{\text{mot}} + \hat{\theta}_{\text{ext,2}} \quad (8)$$

The separation of the combustion pressure by using the extrapolation method is shown in the right part of Figure 4 for different operating points.

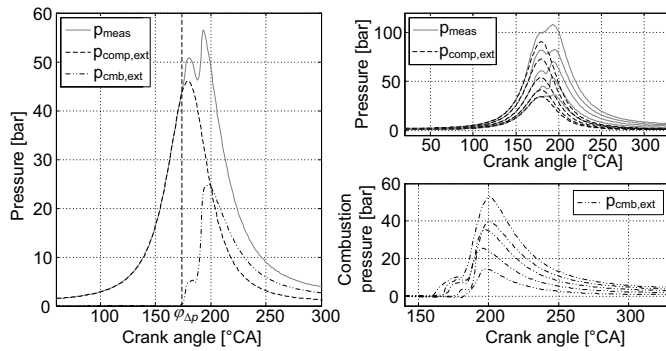


Figure 4: Extrapolation of compression pressure trace

The accuracy of this approach was analyzed by using measurements with the injection shut-off in the first cylinder. Figure 5 shows the extrapolation errors, whereas the area used for adaption is shaded in gray. Again the best results could be obtained when using the identified model for motored pressure as basis function (Eq. 6).

The extrapolated pressure traces serve as basis for the identification of the compression pressure model, whereas the engine parameters intake manifold temperature, EGR rate and in-cylinder gas mass are considered as additional regressors. The determined model for the pressure in motored condition is used as basis function. In order to avoid overfitting the data set is split in 50% training data, 25% validation data and 25% test data [18]. Eq. 9 exemplifies an identified compression pressure model.

$$p_{\text{comp}}(\varphi) = \hat{\theta}_{\text{comp}} \mathbf{Z}_{\text{comp}}^T \quad (9)$$

$$\mathbf{Z}_{\text{comp}} = \begin{bmatrix} p & p T_{\text{IM}}^2 & p m_{\text{gas}} & p p_{\text{IM}} m_{\text{gas}} & p_{-1} T_{\text{IM}} m_{\text{gas}} & p_{-2} p_{\text{IM}}^2 \end{bmatrix}$$

where: $p = p_{\text{mot}}(\varphi)$, $p_{-1} = p_{\text{mot}}(\varphi - 1)$, $p_{-2} = p_{\text{mot}}(\varphi - 2)$

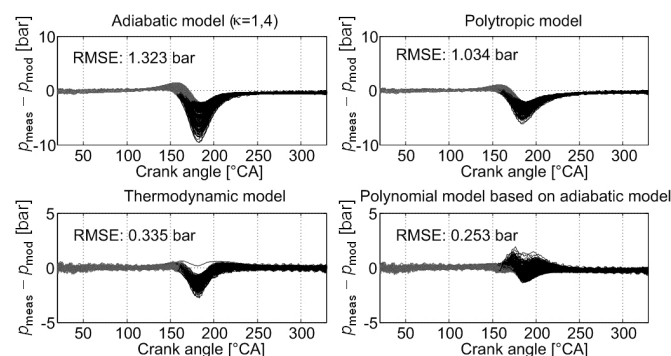


Figure 5: Validation of extrapolation approach (222 operating points, engine PC 1a, cf. Table 4)

Figure 6 shows the errors between modeled and extrapolated compression pressure for test data. To sum up, the advantage of the described method lies in a fast identification and evaluation along with a small number of model parameters.

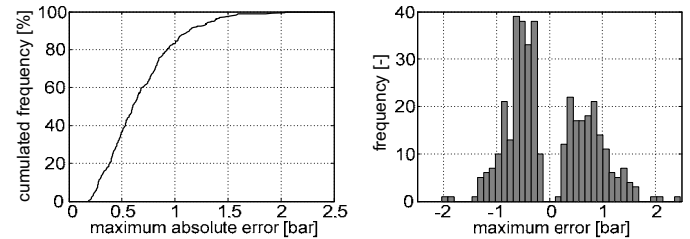


Figure 6: Maximum error between modeled and extrapolated compression pressure for test data (408 operating points, Dataset A, cf. Table 4)

IDENTIFICATION OF IGNITION DELAY – The precondition for modeling the ignition delay is an automated and correct determination of the starts of combustion from measurement data. In general the identification is based on the calculation of the heat release rate [3]. However, the robustness of this approach is often not sufficient due to increasing signal distortions at high engine speeds. Hence a direct identification of the ignition delay from the pressure curve is carried out. The basis is formed through the introduced compression pressure model which allows for an exact determination of the combustion pressure.

Pilot combustion – In the case of two pilot injections a joint combustion is typically observed and hence no exact separation is possible in most cases. Thus only a single contribution of the pilot injections is modeled, whose start of combustion is defined as the position of the first significant energy release. Again the identified model for motored pressure serves as starting point and is adapted to the measured compression curve in the range from φ_{IC} to the first start of injection according to Eq. 7 and 8. The start of the pilot combustion is defined as the crank angle where the combustion pressure exceeds a small threshold, e.g. 0.25 bar (Figure 7).

Main combustion – In order to identify the start of main combustion the curvature $\gamma(\tilde{\varphi})$ of the combustion pressure is calculated and the local maximum in the range after the start of main injection is searched (Eq. 10, Figure 7).

$$\gamma(\tilde{\varphi}) = \frac{\tilde{p}_{\text{cmb}}''(\tilde{\varphi})}{\left[1 + \tilde{p}_{\text{cmb}}'(\tilde{\varphi})^2\right]^{3/2}} \quad (10)$$

$$\text{where } \tilde{\varphi} = \frac{\varphi}{1^\circ\text{CA}}; \tilde{p}_{\text{cmb}}(\tilde{\varphi}) = \frac{p_{\text{cmb}}(\varphi)}{1 \text{ bar}}$$

If no pilot injection was detected, the start of main combustion is also determined by a threshold-comparison.

Figure 7 exemplifies that an accurate compression pressure model is an essential prerequisite for a correct determination of the ignition delay and the pressure contribution of the pilot injection.